

EFFECT OF PASSIVES ON THE PERFORMANCE OF A CENTRIFUGAL COMPRESSOR

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ABSTRACT

The aim of the present experimental investigation is to investigate the feasibility of utilizing passive means to enhance the performance of a centrifugal compressor. The experimental investigation is done with partial shroud made of mild steel fitted on the tip of the rotor blade impeller exit traverse with a pre-calibrated five hole probe, and turbulence generator made with Velcro tape fitted inside the inducer leading edge impeller inlet traverse with a pre-calibrated three hole probe, for three flow coefficients, $\phi = 0.28$ (near design), $\phi = 0.34$ (above design) and $\phi = 0.18$ (below design). From the present experimental investigation, it is presumed that partial shroud on the tip of the blade has valuable impacts in efficiency and energy coefficient, as compared to without partial shroud on the tip of the blade.

KEYWORDS: Centrifugal Compressor, Spacers, Passives & Tips Clearance

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1. INTRODUCTION

The flow field of an impeller exit of a centrifugal compressor is exceedingly unpredictable, three-dimensional and unsteady. A producer for a centrifugal compressor is to have entire understanding on the affecting parameters, there by one can plan for better performance and efficiency. While running, there is a leakage flow, through this tip clearance which deteriorates the performance of the machine. i.e. reduces the pressure ratio, increases the input power for decreasing the efficiency. The main components of a centrifugal compressor are impeller and diffuser. There are many parameters that affect the flow in a centrifugal impeller. The geometric and flow parameters that affect the flow in a centrifugal compressor are: Inlet geometry, Pre-whirl, Inducer geometry, Bends at the inlet, Flow angles, Clearance between impeller blades and stationary shroud, Rotational speed, Diffuser type and shape of the volute casing. Among these, Clearance between the impeller blades and stationary shroud is the most important phenomenon of consideration. The performance of a compressor is inherently deteriorated by different losses that occur at different sections of the compressor stage. They are as follows; shock loss at an inducer inlet, wall friction loss within impeller channels, tip leakage loss, secondary flow losses, mixing loss at diffuser inlet, wall friction loss within a diffuser, sudden expansion loss at a scroll inlet, wall friction loss within a scroll. The effect of tip clearance on the performance of a centrifugal compressor has been studied by many persons. Ishida and Senoo (1981) measured pressure distribution along the shroud at seven different tip clearances. They concluded that the pressure loss due to tip clearance is proportional to the pressure rise due to deceleration of relative velocity. They also observed that the change in input power due to a change of tip clearance is related to the effective blockage at the impeller exit. Ishida and Senoo (1986) modified their own theory on tip clearance loss of centrifugal impellers as a) Efficiency drop due to tip clearance of high pressure ratio compressors is less than that of

low pressure ratio compressors, if the tip clearance ratios at the impeller exit are equal. b) The magnitude of clearance loss becomes smaller as the flow rate is reduced and also at a reduced shaft speed in cases of high pressure ratio compressors. Freeman (1985) suggested that 1% increase in tip clearance produced by increasing the casing diameter will give approximately 1.4% loss in efficiency. Schumann et al. (1987) tested a centrifugal impeller with a vane less diffuser for four impeller area ratios from 2.322 to 2.345. The impeller was initially designed for a pressure ratio of approximately 5.5 and a mass flow rate of 0.959 Kg/s. For each area ratio, a series of impeller exit axial clearances was also tested. They found that impeller efficiency decreases by 0.4 point for every 1% increase in exit clearance. Smith and Cumpsty (1984) have shown a 23 percent drop in maximum pressure rise and a 15 percent increase in flow coefficient at stall in a large, low speed compressor as the tip clearance was increased from 1 to 6 percent of chord.

2. EXPERIMENTAL SETUP

The major geometrical details of the impeller are total pressure rise, ΔP : 300 mm WG, Speed of rotation, N : 2000 rpm, Impeller inlet diameter at inducer tip, d_{it} : 300 mm, Impeller inlet diameter at inducer hub, d_{ih} : 160 mm, Impeller outer diameter, d_2 : 500 mm, Blade angle at exit: (a) At hub: 75° , (b) At mean section: 90° , (c) At shroud: 105° , Blade angle at inlet inducer hub, β_{ih} : 53° , Blade angle at inducer tip, β_{it} : 35° , Blade width at the exit, b : 34.7 mm, No. of blades of the impeller, N_b : 16. All the angles are with respect to the tangential direction.

A schematic design of the centrifugal compressor is appeared in Figure 1. The experimental set up comprises of basically a diffusive impeller driven by a 5 kW D.C. motor with an appraised speed of 2000 rpm. The D.C. motor is specifically directly coupled to the shaft of the impeller. The primary parts of the compressor are suction channel (duct), impeller, vane-less diffuser shaped by the front and back walls of the casing and volute casing and a delivery pipe with a throttle at its outlet and nozzle at the inlet.

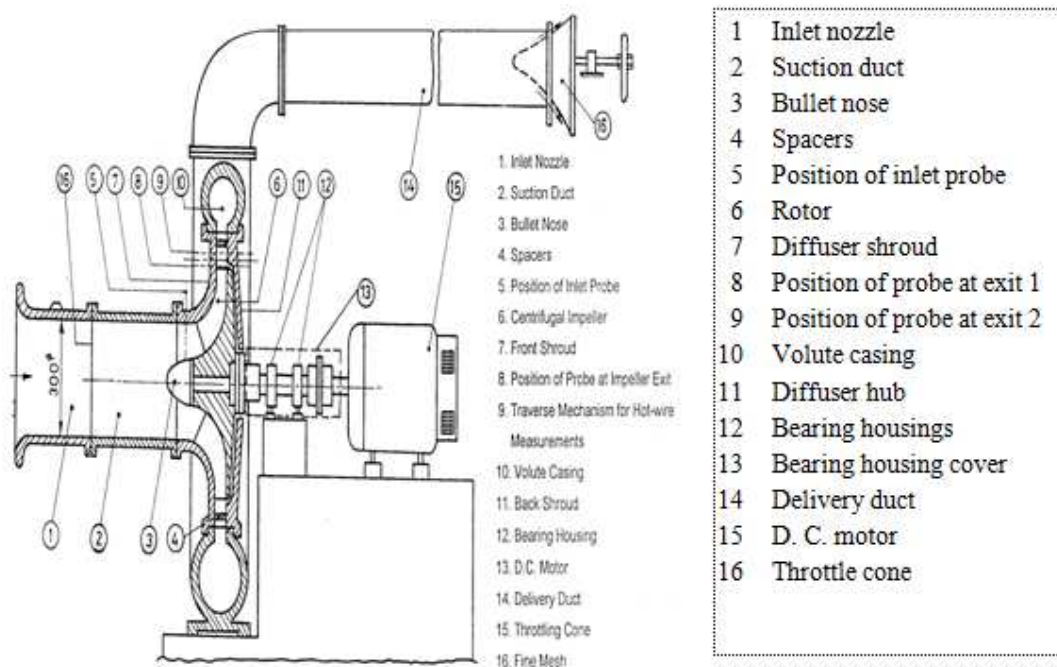


Figure 1: Schematic Diagram of Centrifugal Compressor

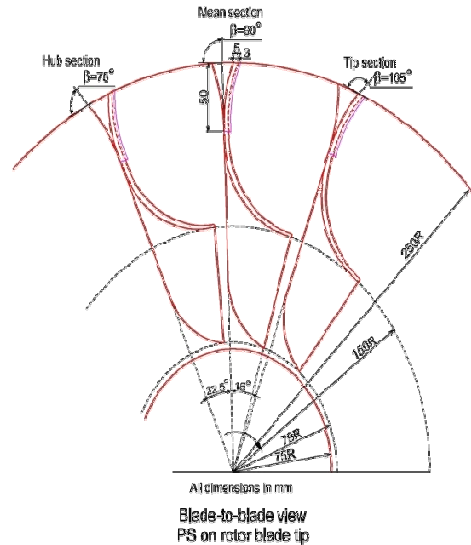


Figure 2: Blade-To-Blade View of the Impeller with PS on Rotor Blade Tip

The objective of the present investigation is to study the effect of partial shrouds and turbulence generator on performance of a centrifugal compressor and to achieve increased operating range and efficiency of a centrifugal compressor. The effect of tip clearance on the performance characteristics the different flow coefficient and different tip clearances is studied. The tip clearance is varied by using spacers. The volume flow rate is varied with the help of throttling device to conduct the performance test on centrifugal compressor. The flow is a three dimensional phenomenon comprised of the complex interactions between the tip leakage vortex, the turbulent end wall boundary layers, and often shock waves distorted by Coriolis and rotational effects.

Partial shrouds are made of stainless steel of 0.1 mm thickness. The stainless steel sheet is sliced to the state of rectangle bits of 50 mm x 5 mm measure. These rectangle pieces are fixed to the tip of the blades by using araldite. The configurations tested (basic configuration without partial shroud and configuration with partial shroud) are shown in Figure 3. The blade to-blade edge see demonstrating the partial shroud on the rotor tip is appeared in Figure 2.



Figure 3: Blade-To-Blade View of the Impeller with PS on Rotor Blade Tip

Turbulence generator: Turbulence generator is utilized as a stumbling gadget in this experiment for the creation of turbulent boundary layers. A turbulence generator (TG) is connected to within the suction conduit upstream of the rotor. The turbulence generator (TG) is made of Velcro tape. The turbulence generator is appended to within the suction conduit. The hub width of turbulence generator is 10 mm, length 942.5 mm, thickness 2 mm. A mild steel ring of 16 mm axial width, 3 mm thickness and 930 mm length was utilized to attach the turbulence generator to the suction conduit. A layer of Araldite of small thickness was utilized between the tape and the casing wall to join the tape utilizing Araldite. The ring

applied uniform circumferential pressure all around the tape. The additional araldite left the crevice between the suction duct wall and the turbulence generator. It was analyzed that the time taken for the Araldite to solidify was limited and the thickness of solidified Araldite was higher uniform. Additionally there was no sag of turbulence generator, and there was no gap between turbulence generator and suction duct the turbulence generator as shown in figure 4.



Figure 4: Without and with Turbulence Generator (TG) is Attached to Inside of the Suction Duct 15 mm Upstream of the Rotor.

3. INSTRUMENTATION

The performance of the compressor was controlled by the adjustment in the static pressure over the compressor. The static pressure on the suction pipe and delivery pipe were measured utilizing a scanning box (Model FC091-3) and micro manometer (demonstrate FCO12) made by M/s Furness Control Ltd., Bexhill, UK. The scanning box contained 20 valves, which are numbered successively. The pressures to be measured were associated with the numbered inputs. Pressure inputs are perused in arrangement by utilizing the micro-manometer. The micro-manometer is a delicate differential pressure measuring unit, capable of reading air pressures from 0.01 mm to 2000 mm water gage (WG). It would react to weight contributions up to 50 Hz. Be that as it may, the time consistent potentiometer can be utilized to normal the pressure variances.

The speed of the centrifugal compressor was measured utilized a non-contact type digital tachometer. Four interconnected static pressure tapings on the inlet ringer mouth casing wall at the throat segment were utilized to decide the inlet velocity. The bell mouth zone and the volume flow were computed utilizing a reasonable estimation of coefficient of discharge for the bell mouth. A D.C. motor with a different exciter is utilized to drive the rotor of the centrifugal compressor. The input power was measured by mean of voltmeters and ammeters associated independently for field and armature supplies. A reasonable estimation of motor efficiency was utilized to get the rotor input power. The Micro manometer is shown in Figure 5.



Figure 5: Micro Manometer

4. RESULTS AND DISCUSSIONS

Performance Characteristics: The results of the present investigation are presented and discussed in this section. The performance of the compressor in terms of ψ vs. ϕ and η vs. ϕ are presented. The non-dimensional parameters are defined as follows:

$$\text{Flow coefficient, } \phi = \frac{V}{\pi d_2 b_2 U_2} = \frac{c_{2r}}{U_2}$$

Where V = Volume flow rate (m^3/s)

d_2 = Rotor tip diameter (m)

b_2 = Rotor blade width at exit (m)

c_{2r} = Radial velocity at rotor exit (m/s)

U_2 = Rotor tip speed (m/s)

The specific work W for the centrifugal compressor is expressed non-dimensionally as the energy coefficient, ψ

$$\psi = \frac{2W}{U_2^2} \quad \text{Where } W = \text{Specific work } (\text{m}^2/\text{s}^2)$$

Where Specific work W is obtained as the difference of specific total energy between inlet and exit of the impeller. For incompressible flow, the specific work is given by,

$$W = \frac{P_d - P_s}{\rho} + \frac{c_d^2 - c_s^2}{2} + g\Delta Z$$

Where P_d and P_s are the average static pressure measured at the exit and inlet conditions. C_d and C_s are air velocities at the exit and inlet sections. ΔZ is the geodetic level difference between delivery and suction flanges and ρ is the density of air. The density of the air is calculated from atmospheric pressure and temperature.

$$\text{Power coefficient, } \gamma = \frac{2N_c}{\rho A U_2^3} = \frac{2\eta_m EI}{\rho A U_2^3}$$

Where

N_c = Coupling power (Watts) = $\mu_m [V_A I_A + V_F I_F]$

A = Suction duct area (m^2)

E = Motor voltage (Volts)

I = Motor current (Amps)

η_m = Motor efficiency

U_2 = Tip speed of impeller

V_A = Armature voltage, Volts

V_F = Field voltage, Volts

I_F = Field current, Amps

I_A = Armature current, Amps

$$\text{Compressor efficiency, } \eta = \frac{\rho V W}{N_c}$$

The performance characteristics of a centrifugal compressor are evaluated at a constant speed of 2000 rpm with 2%, 4%, 6% and 8% tip clearance for normal (without passive means) centrifugal compressor by varying flow coefficients. The compressor characteristic curves energy coefficient and efficiency against flow coefficient at different tip clearances is shown in Figure 6 & 7. In the stage of normal centrifugal compressor is shown maximum efficiency of 46.1% with 2% tip clearance as compared to other tip clearances. Using the micro manometer the static pressure between the entry and exit of the compressor difference is measured, at different tip clearances. The figure depicts a negligible difference in efficiency for flow coefficients up to 0.3, for all tip clearances. At flow rates higher than this, the difference in efficiency is appreciable, the peak efficiency being lower for higher tip clearance and vice versa. This can be attributed to the increase in tip losses with the increase in the tip clearance. The reduction in efficiency is due to the reduction in energy transfer for higher tip clearance.

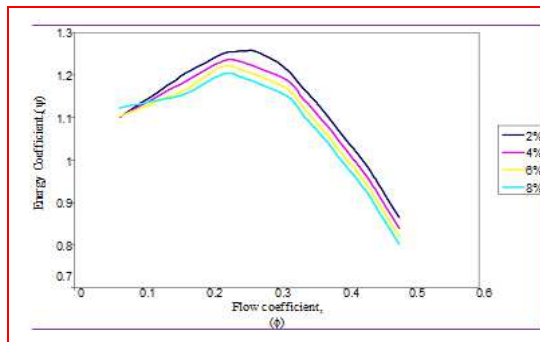


Figure 6: Energy Coefficient vs. Flow Coefficient

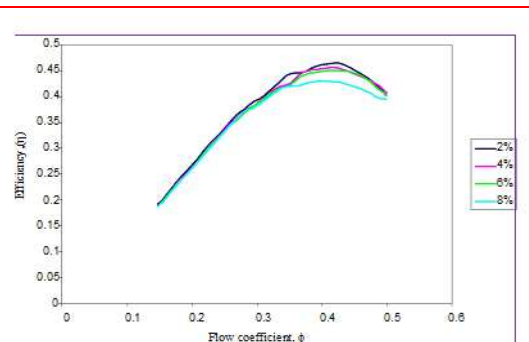


Figure 7: Efficiency vs. Flow Coefficient

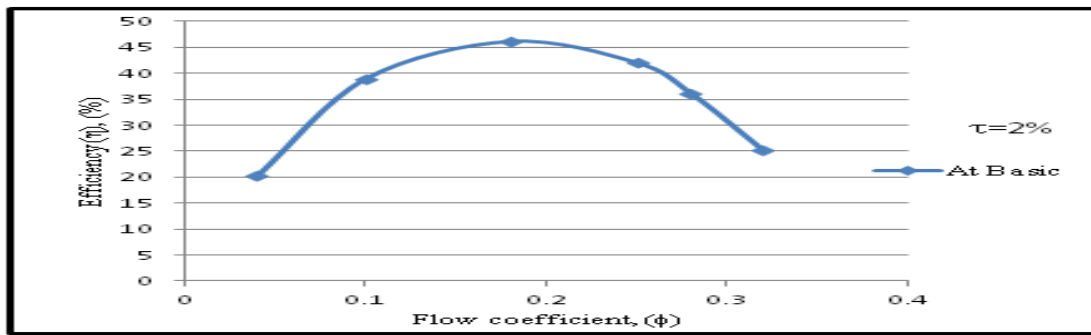


Figure 8: Efficiency vs. Flow Coefficient at 2% of Tip Clearance

The same performance characteristics of a centrifugal compressor are evaluated for this paper along with turbulence generator. the turbulence generator is placed at 5mm, 10mm, 15mm, 20mm, 25mm and 30mm upstream of inducer leading edge and performance of a compressor is evaluated for three different tip clearances (2.2%, 5.1% and 7.9%), at different flow coefficients, $\phi = 0.18$ (below design flow coefficient), $\phi = 0.28$ (design flow coefficient) and $\phi = 0.34$ (above design flow coefficient). The turbulence generator with and without placed at different positions of inducer leading edge is shown in Figure 9 & 10. The effect of turbulence generator on the performance of a centrifugal compressor is nominal at lower tip clearances. However stable operating range of compressor small increases at the higher values of tip clearance. The figure shows the higher efficiency configurations with turbulence generator on the casing at 15 mm upstream of inducer tip leading edge with turbulence generator as compare to other positions.

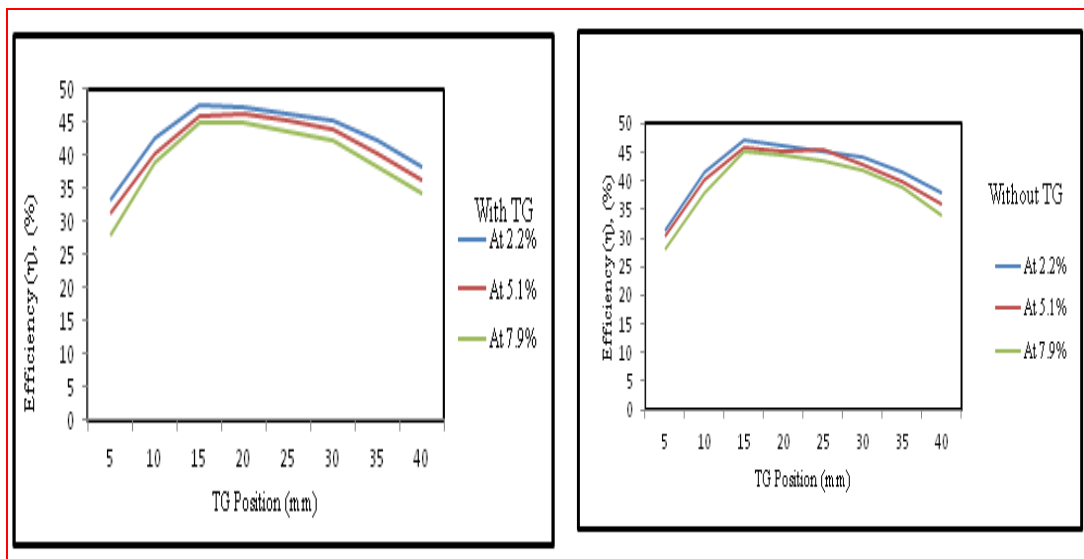


Figure 9: Efficiency vs. Turbulence Generator Positions with TG

Figure 10: Efficiency vs. Turbulence Generator Positions without TG.

The performance characteristic of a centrifugal compressor with configuration TG + PS is shown maximum efficiency of 50.0% at 2.2% tip clearance as compared to all other configurations.

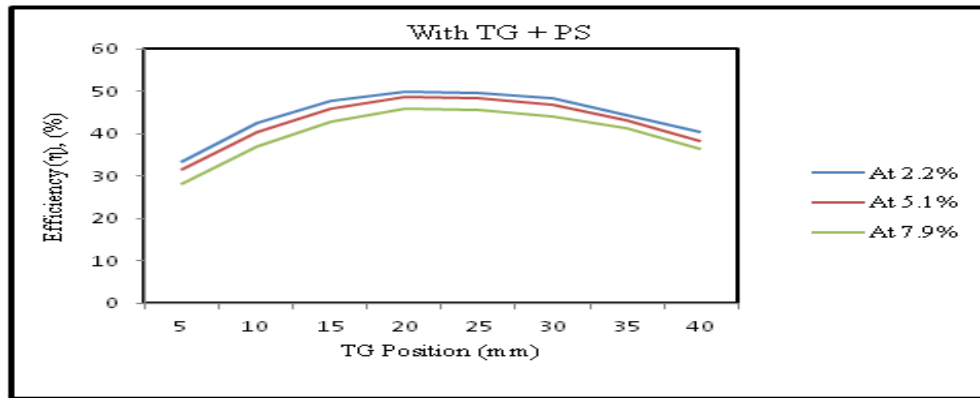


Figure 11: Efficiency vs. Turbulence Generator Positions with TG+PS

The performance characteristics of a centrifugal compressor are evaluated in this stage also by varying tip clearance from 1% to 8%, with and without partial shroud placed on the tip of the blade is shown in Figure 12. The compressor efficiency is reduced as the tip clearance is increased. This can be attributed to the increase in tip leakage flow with the increase in the tip clearance. Configurations with partial shroud clearly shows that the higher efficiencies as compare to configuration without partial shroud. Because of the partial shroud reducing the tip gap and the extension of the partial shroud pressure surface side, with this the tip leakage flow has to travel a longer distance before interacting with the main flow. Figure shows that the compressor performance is maximum at tip clearance 2.2%, in the range from 2% to 5%, tip clearance 5.1% is shows best performance for the next range from 5% to 7% and tip clearance 7.9% is shows best performance for the last range from 7% to 9% tip clearance is observed.

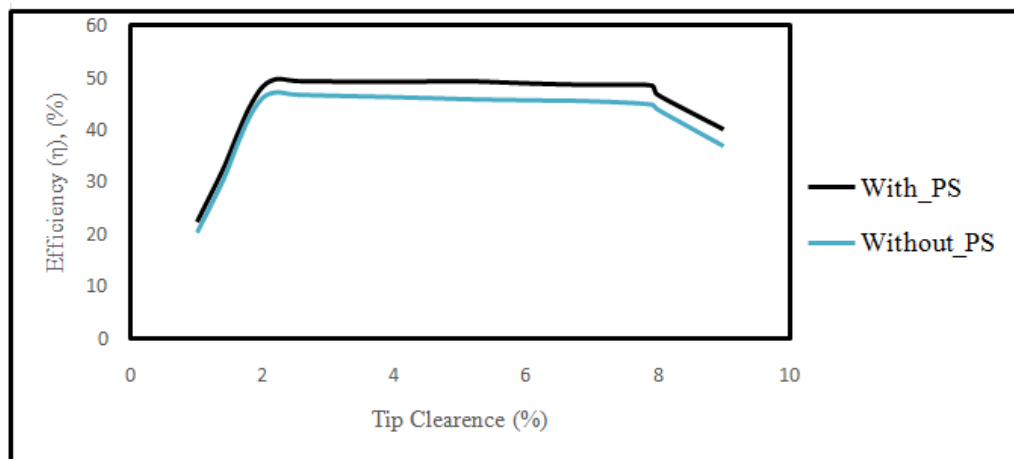


Figure 12: Efficiency vs. Tip Clearance with and without PS

CONCLUSIONS

The following major conclusions are drawn from this present experimental and computational investigation.

- The present experimental investigation reveals that the configuration at normal compressor has show 46.1% of efficiency at 2% (0.7 mm) tip clearance for design flow coefficient.
- Turbulence generator on the casing at 15 mm upstream of inducer tip leading edge has shown 47.1% of efficiency

at 2.2% of tip clearance. Turbulence generator (TG) has nominal effects on the energy coefficient and efficiency of the compressor. However the operating range of the compressor is substantially increased.

- The compressor efficiency is slightly increased by increasing tip clearance from 2% to 2.2% of tip clearance.
- Partial shroud on tip of the blade increased the efficiency of compressor by 49.8% at 2.2% of tip clearance. Configurations with partial shroud (PS) shows higher energy coefficient and efficiency compared to the basic configuration at all the values of tip clearance.
- Combination of partial shroud on tip of the blade and turbulence generator on the casing at 15 mm upstream of inducer tip leading edge, the efficiency of compressor has increased to 50% from 49.8% as compared with only partial shroud.

NOMENCLATURE

b	Distance between the shroud and hub at the rotor exit (m)
C_d	Velocity in delivery duct (m/s)
C_s	Velocity in suction duct (m/s)
d	Rotor diameter (m)
N	Rotational speed of rotor (rpm)
N_c	Coupling power (Watt)
N_{sh}	Shape number = $N\sqrt{V}/W^{3/4}$
P_s	Static pressure (Pa)
P_o	Total pressure (Pa)
p_d	Delivery pressure (Pa)
p_s	Suction pressure (Pa)
t	Clearance of the rotor blade (m)
U_2	Rotor tip speed = $(\pi d_2 N/60)$ (m/s)
V	Volume flow rate (m^3/s)
W	Specific work (m^2/s^2)
ϕ	Flow coefficient (defined in the text)
γ	Power coefficient
η	Efficiency (defined in the text)
ψ	Energy coefficient (defined in the text)
ψ_o	Total pressure coefficient = $2P_o/U_2^2$
ψ_s	Static pressure coefficient = $2P_s/U_2^2$
ρ	Density of air (kg/m^3)
τ	Relative tip clearance = (t/b_2)
Superscript	
-	Mass averaged value
Subscript	
2	Tip

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